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Investigation of a Gas-Driven Jet Pump for Rocket Engines

by

D. L. Crabtree

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JET PROPULSION CENTER
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LAFAYETTE, INDIANA

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AND

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Final Report

INVESTIGATION OF A GAS-DRIVEN JET PUMP FOR ROCKET ENGINES

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I. INTRODUCTION

The object of the investigation was: (1) to establish the analytical and experimental background necessary for describing a gas-driven jet pump, and (2) to study the application of that type of jet pump to the rocket engine system. The investigation of the gas-driven jet pump was initiated in 1956, at the Jet Propulsion Center, Purdue University.

Figure 1 presents a block diagram of the jet pump system employed in the investigation. A liquid and a gas at high pressure (typically 500 psi to 600 psi) are introduced into a bi-fluid injector; the aforementioned components of the resulting two-phase flow are termed the drive liquid and the drive gas, respectively. The drive liquid flow rate is 10 to 20 times as large (by weight) as the drive gas flow rate. The ratio of the drive gas flow rate to the drive liquid flow rate is termed the mixture ratio of the drive nozzle and is expressed as a percentage. The two drive fluids are injected into the entrance of a converging-diverging nozzle, termed the drive nozzle, where they are mixed to form a high pressure (typically 500 psi), two-phase mixture. The expansion of the two-phase mixture in the drive nozzle, from the high pressure to a low pressure (typically ambient), produces a high velocity two-phase jet (500 fps to 800 fps) at the exit of the drive nozzle, termed the drive jet. The latter then enters an induction-acceleration device, termed the mixer, wherein the liquid in the drive jet impinges on the suction liquid, the liquid to be pumped, and transfers momentum to the suction liquid. The mixture of drive liquid, suction liquid, and some drive gas entrained in the liquid, moving at a velocity of approximately 350 fps,

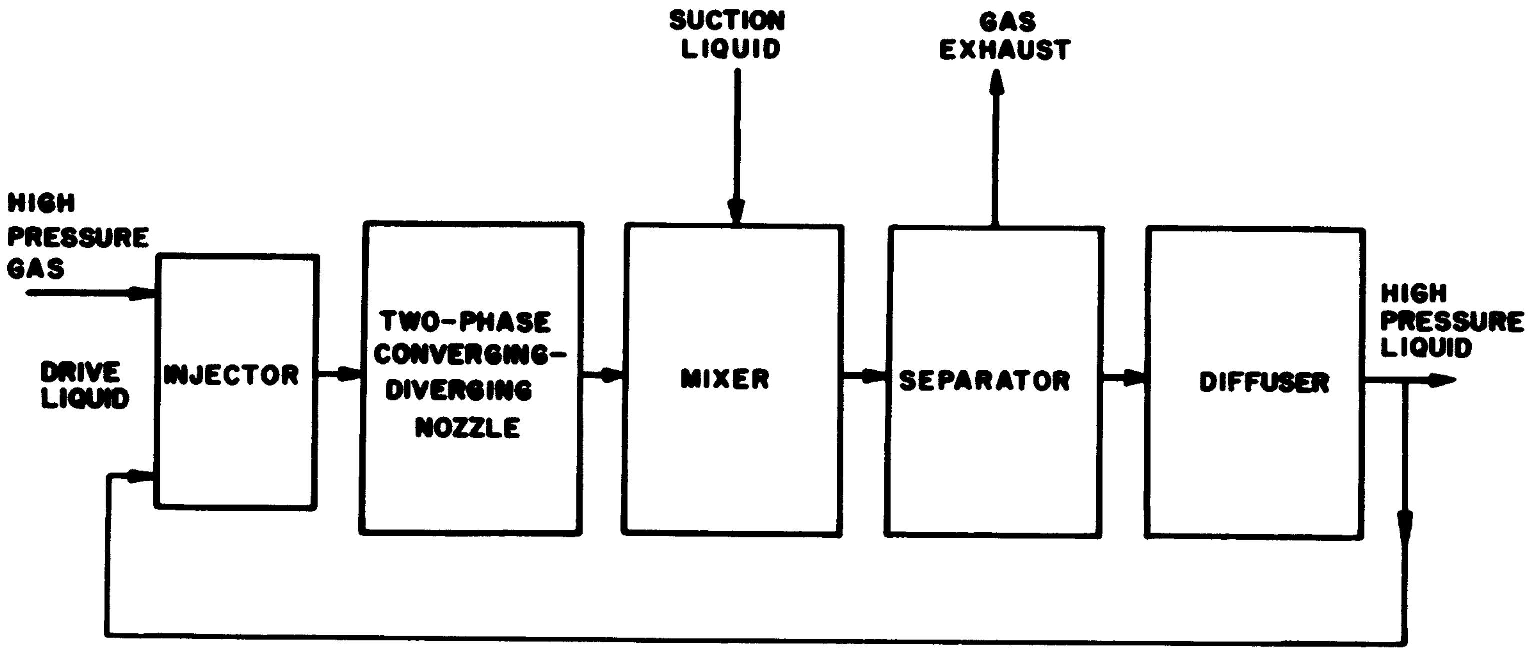

Fig. I. Block Diagram of Gas-Driven Jet Pump enters a device termed the separator where most of the entrained drive gas is removed from the liquid by a process of centrifugal separation. The essentially gas-free liquid, still at a high velocity, enters the diffuser wherein the dynamic pressure of the flowing fluid is converted to static pressure; the flow is then discharged from the jet pump. A portion of the discharged liquid is cycled back to the bi-fluid injector and comprises the afore-mentioned drive liquid.

II. ANALYSIS OF THE CHARACTERISTICS OF A GAS-DRIVEN JET PUMP

The three principal characteristics employed for evaluating the feasibility of applying the jet pump to a rocket engine system were: (1) the theoretical performance and the gas consumption, (2) the capability of pumping common rocket propellants, and (3) the estimated weight of the jet pump.

(a) Theoretical Performance and Gas Consumption

The gas consumption of a pumping system is defined as the ratio of the mass flow rate of gas used in the jet pump to the net mass flow rate of liquid discharged from the system. From the standpoint of weight limitations imposed on a flight vehicle employing a gas-driven jet pump, it is generally desirable to minimize the gas consumption.

The gas consumption of a gas-driven jet pump can be approximated from the following relationship (1).* Thus,

$$\frac{\dot{M}_G}{\dot{M}_D} = 2\left[\frac{1 + \sqrt{1 - K^2}}{K^2}\right]\left[\frac{p_D - p_S}{\ln p_N/p_S}\right]\left[\frac{W}{R\ t_N\ \rho_L}\right] \tag{1}$$

where $\dot{M}_G$ = the mass flow rate of drive gas,

$\dot{M}_D$ = the mass flow rate of the discharged liquid,

K = the momentum recovery factor of the jet pump,

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* Numbers in parentheses refer to references listed in the back of the report.

p_D = the discharge pressure,

p_S = the suction liquid pressure in the mixer,

p_N = the pressure at the injector face in the drive nozzle,

W = the molecular weight of the drive gas,

ρ_L = the density of the liquid, and

R = the universal gas constant.

The magnitude of the first bracketed expression in Equation 1, termed the loss factor, depends on the pump losses. In the analysis of the jet pump the momentum recovery factor, K, of the jet pump is equal to the product of the momentum recovery associated with each of the components.* Thus,

$$K = K_1 K_2 K_3 K_4 \sqrt{K_D} \tag{2}$$

where K = the momentum recovery factor of the jet pump,

K_1 = the velocity recovery factor of the drive nozzle defined to be the ratio of the measured effective nozzle exit velocity to a calculated isentropic exit velocity,

K_2 = the momentum recovery factor of the mixer defined to be the ratio of the momentum of the liquid entering the separator to the momentum of the liquids entering the mixer,

K_3 = the mass recovery factor defined to be the ratio of the flow rate of liquid entering the diffuser to the flow rate of liquid entering the mixer,

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* An analysis based on momentum was chosen instead of one based on energy since the analysis based on momentum proved to be more amenable to experiment.

K_4 = the velocity recovery factor of the separator defined to be the ratio of the velocity of the liquid leaving the separator to the velocity of the liquid entering the separator, and

K_D = the efficiency of the diffuser defined to be the ratio of the static pressure rise in the diffuser to the dynamic pressure of the liquid entering the diffuser.

Figure 2 presents the loss factor as a function of the momentum recovery factor, K. Values of K less than 0.4 are of little practical interest and the value of K = 1.0, represents the largest thermodynamically possible. It can be seen from Fig. 2 that even relatively small losses result in values of the gas consumption which are many times the minimum thermodynamically possible. If K_1, K_2, K_3, K_4, and K_D are each equal to 0.9, the momentum recovery factor of the jet pump is approximately 0.6. The corresponding loss factor is 5.0; and the gas consumption is 5 times the thermodynamic minimum.

The second bracketed expression in Equation 1 is termed the <u>pressure factor</u> and is a measure of the effects of the discharge pressure, p_D, the suction pressure, p_S, and the nozzle pressure, p_N, on the gas consumption. For any practical jet pump system the drive gas would be generated in a bootstrap operation, and, therefore, the nozzle pressure would be less than the discharge pressure. The gas consumption would be from 3 per cent to 10 per cent larger if the pressure in the drive nozzle were 100 psi less than the value of the discharge pressure compared to the gas consumption if the nozzle pressure and discharge pressure were equal. Hence, bootstrap operation of a jet pump does not introduce an intolerable increase in the gas consumption.

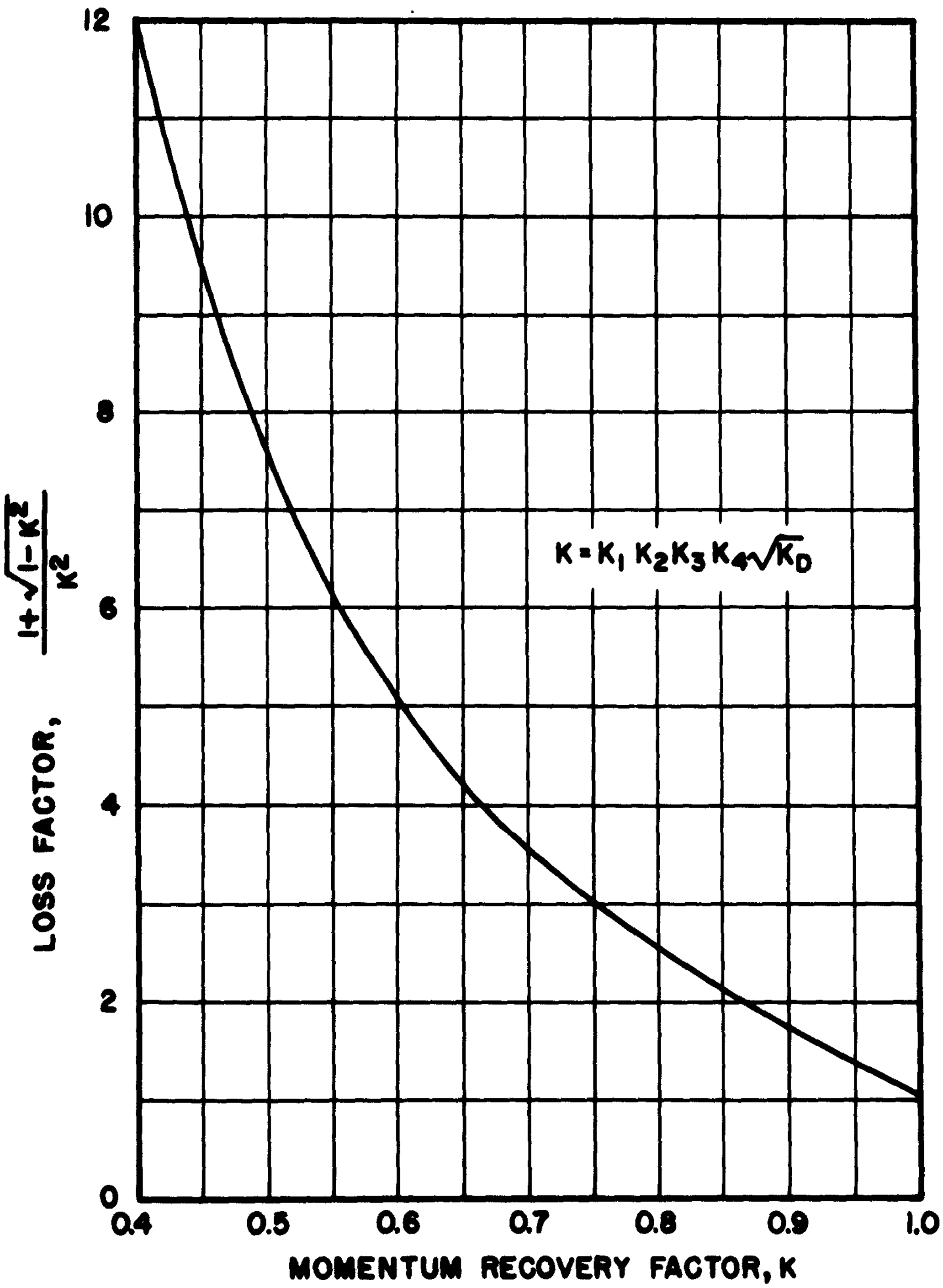


Fig. 2. Loss Factor as a Function of the Momentum Recovery Factor

The third bracketed expression in Equation 1 presents the effect of the density of the liquid, ρ_L, the molecular weight of the drive gas, W, and the equilibrium temperature of the two-phase mixture at the entrance to the nozzle, t_N, on the gas consumption. Theoretically, Equation 1 indicates that the gas consumption of a jet pump will be minimized for fluid property effects by using a drive gas of low molecular weight to pump a liquid with a large density at the highest feasible temperature of the drive liquid flowing through the drive nozzle.

(b) Pumping Propellants with a Jet Pump

The three important characteristics of a propellant to be considered in determining the possible application of a jet pump to a rocket engine system are: (1) density, (2) vapor pressure, and (3) the potential for thermal decomposition. The influence of density of the propellant on the gas consumption was discussed in the last paragraph of Section II(a).

If the magnitude of the vapor pressure of a propellant being pumped is comparable to the pressures encountered during the expansion of the two-phase mixture in the drive nozzle, the propellant may vaporize during the expansion of the two-phase flow in the drive nozzle, and an excessive quantity of propellant vapor will be exhausted from the system. In some cases, for example, with cryogenic propellants, the loss of vaporized propellant may be as important a criterion of the system performance as the gas consumption (1) (2) (3).

In a practical jet pump system the drive gas, which is at high pressure, will be supplied by a gas generator wherein either a monopropellant is decomposed or bipropellants are burned. The gas temperature may range from 1000 F to 2000 F, and the internal energy associated with the hot drive gas could cause undesirable thermal decomposition of the pumped propellant in the drive nozzle. Two possible examples of the foregoing situation arise when hydrazine and hydrogen peroxide are pumped with their own decomposition products.

Figure 3 presents the theoretical gas consumption and vapor loss as a function of the discharge pressure for hydrazine pumped by its own decomposition products, for two values of momentum recovery factor: $K = 0.8$ and $K = 0.6$. It can be seen from Fig. 3 that the gas consumption and vapor loss approximately double as the momentum recovery factor decreases from 0.8 to 0.6. Although the afore-mentioned curves apply to hydrazine they may also be regarded as typical for pumping JP-5.

Figure 4 presents the theoretical gas consumption and vapor loss as a function of the discharge pressure, p_D, for liquid fluorine and liquid oxygen using gaseous helium as the drive gas. The calculations assume a momentum recovery of $K = 0.8$. The curves indicate that the average vapor loss would exceed the gas consumption by approximately 50 per cent. The lower curves of Fig. 4 assume that a condenser is utilized in the gas exhaust system to reduce the vapor loss; the practibility of employing such a condenser system has, however, not been evaluated.

(c) Weight of the Gas-Driven Jet Pump

Potentially, the gas-driven jet pump should be a light weight device. Except for the components of the pump requiring thick sections because of complex contours, such as the drive nozzle and diffuser, the walls of most of the components need only be thick enough to withstand the internal pressure. If several pumps are arranged in an array for accomplishing a desired discharge flow rate, the additional weight of the manifolding must

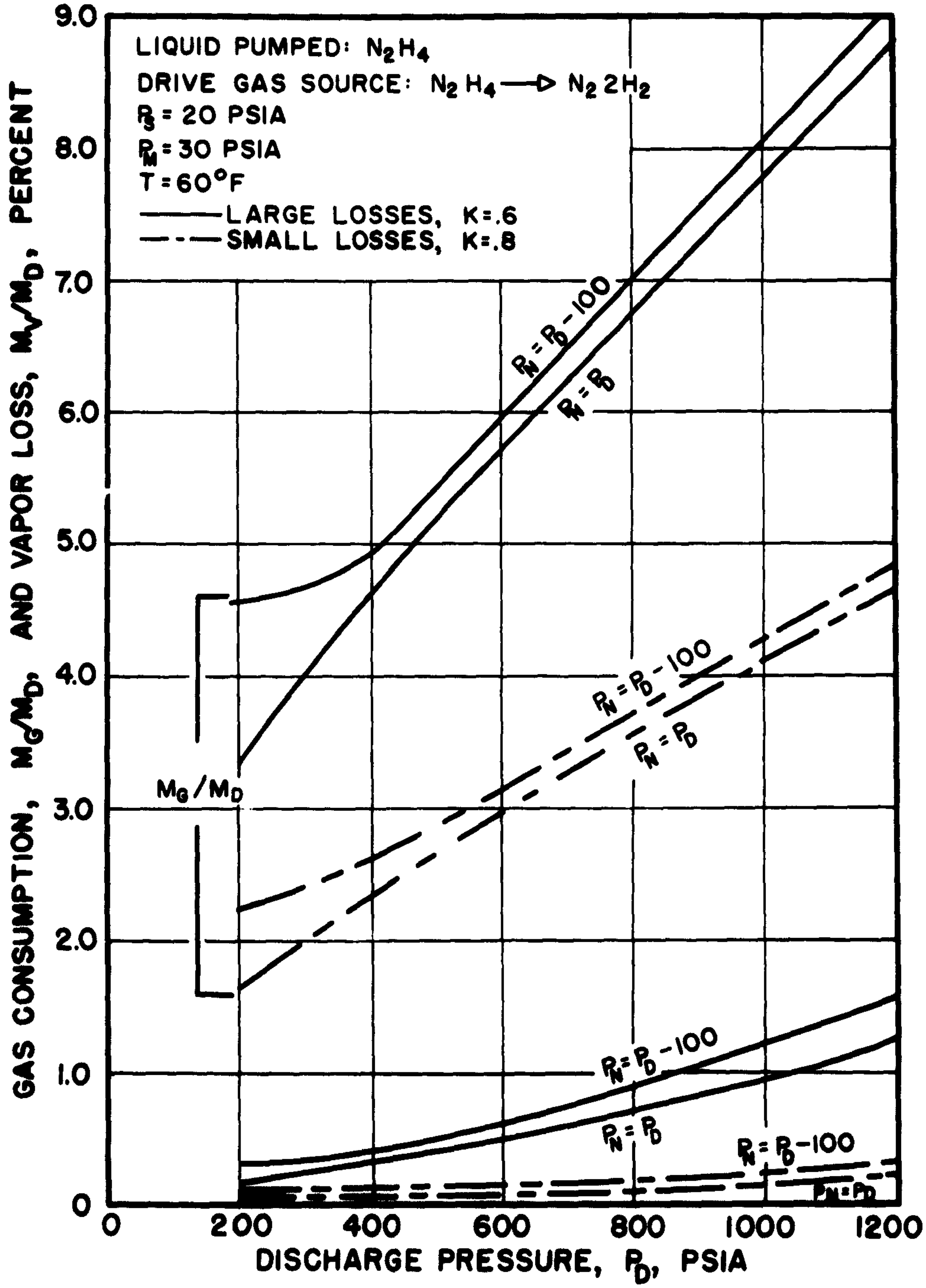


Fig. 3. Theoretical Gas Consumption and Vapor Loss as a Function of the Discharge Pressure for Hydrazine Pumped by the Decomposition Products of Hydrazine

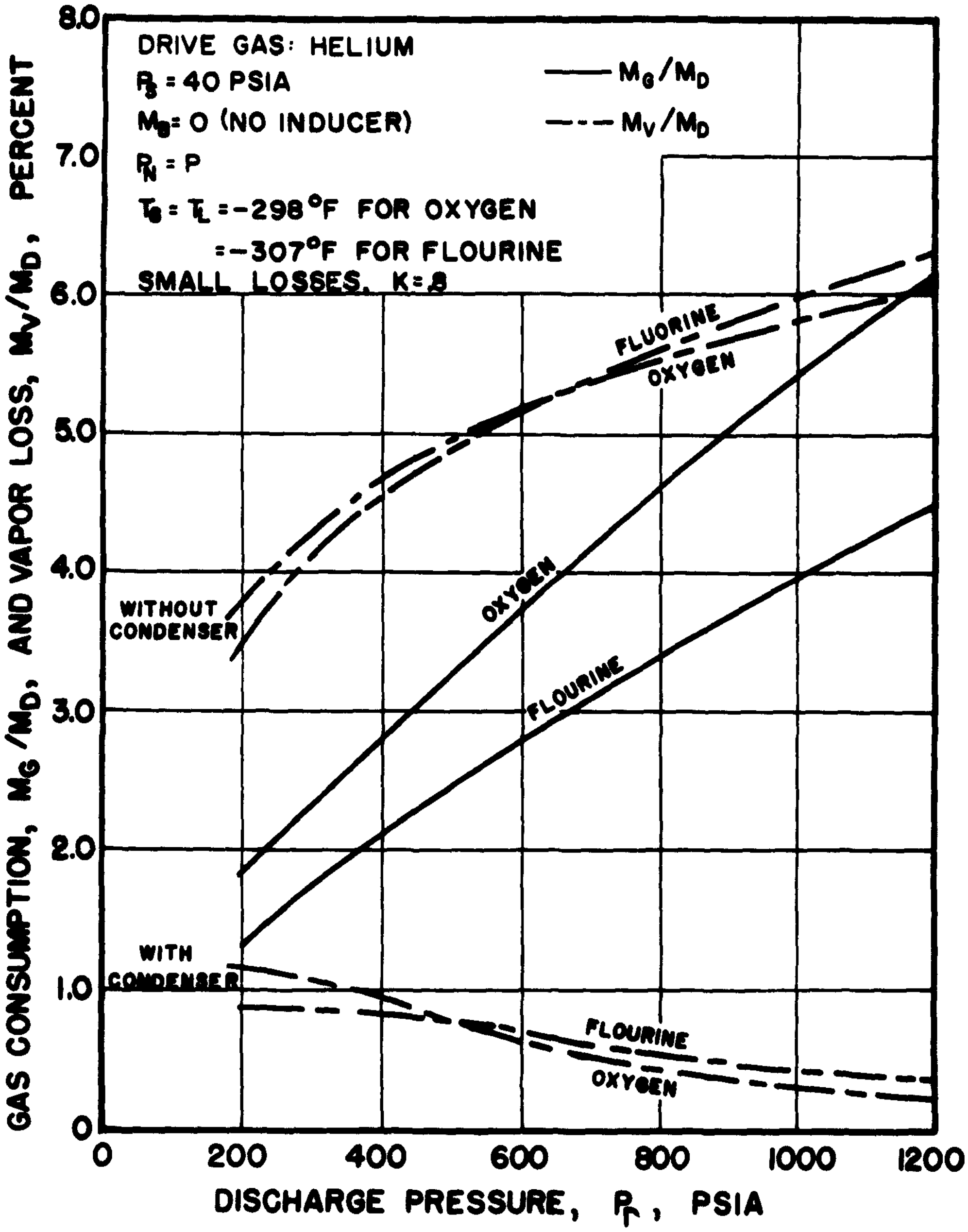


Fig. 4. Theoretical Gas Consumption and Vapor Loss as a Function of Discharge Pressure for Liquid Fluorine and Liquid Oxygen Pumped by Gaseous Helium

be considered. It is estimated that the specific weight of a gas-driven jet pump would be approximately 0.3 to 0.4 lb per pound of propellant pumped per second; this compares favorably with turbopump designs (1).

Figure 5 illustrates schematically the installation of a gas-driven jet pump in a rocket engine system.

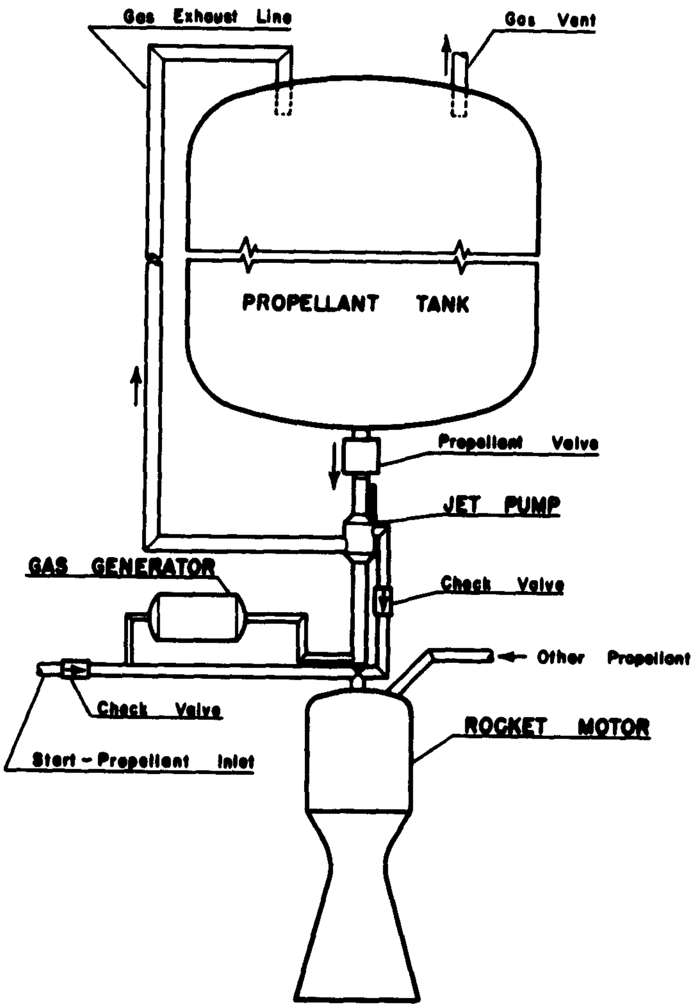


Fig. 5. Installation of a Gas-Driven Jet Pump in a Rocket Engine

III. ANALYTICAL AND EXPERIMENTAL INVESTIGATION OF THE COMPONENT PROCESSES

The five principal processes occurring in the gas-driven jet pump cycle are: (1) the atomization of the drive liquid by the bi-fluid injector to produce a two-phase mixture at the entrance of the drive nozzle, (2) the expansion of the two-phase mixture in the drive nozzle, (3) the transfer of momentum from the drive liquid to the suction liquid in the mixer, (4) the separation of the drive gas entrained in the liquid in the separator, and (5) the recovery of the liquid and the conversion of its dynamic pressure into static pressure in the diffuser.

The analytical investigation of the above component processes was directed toward the following: (1) determining which of several possible methods for accomplishing a process would yield the maximum recovery of momentum, and (2) establishing an analytical description for that method.

The experimental investigation of the component processes was directed toward determining the following: (1) the optimum geometric configuration of the components, and (2) the flow characteristics of the component processes.

(a) The Injector

An ideal bi-fluid injector for a drive nozzle would perform the following functions: (1) distribute the drive liquid and the drive gas uniformly across the entrance of the drive nozzle in the form of a two-phase mixture, and (2) disperse and atomize the drive liquid with the minimum loss in total pressure for each fluid. The formation of the two-phase mixture and the dispersion and atomization of the drive liquid is most readily achieved by increasing the differential pressure with which the drive gas is injected. Large differential pressures, however, lead to increased total pressure losses for both the gas and the liquid and require excessive discharge pressures from the pump, conditions which are incompatible with achieving a low value of gas consumption.

Figure 6 presents photographs of three configurations of bi-fluid injectors employed in the experimental investigation. The three configurations represent (A) twin-fluid atomization, (B) pre-mixing, and (C) parallel streams.

The experimental results indicated that the velocity recovery factor for the drive nozzle was insensitive to the injector configuration, provided that the streams of drive liquid did not flow into the throat region of the drive nozzle before they disintegrated into droplets. The size of the droplets at the exit from the drive nozzle was influenced by the injector configurations (4), i.e., varying inversely to the size of the droplets present in the converging section.

(b) The Drive Nozzle

There are three limiting cases pertinent to the two-phase flow of a gas and liquid in the drive nozzle that can be investigated analytically in a simple manner (1). They are: (1) isentropic flow with no internal temperature difference (infinite heat transfer coefficient between phases), (2) isentropic flow with no internal heat transfer, and (3) separate expansion of the two fluids and then mixing. Figure 7 presents the theoretical exit velocities from the drive nozzle as a function of the mixture ratio, for the three afore-mentioned limiting cases, for two-phase mixtures of air and water. It can be seen from Fig. 7 that an expansion occurring with no internal temperature difference between the phases will result in

Fig. 6. Three Configurations of Bi-Fluid Injectors for Drive Nozzles

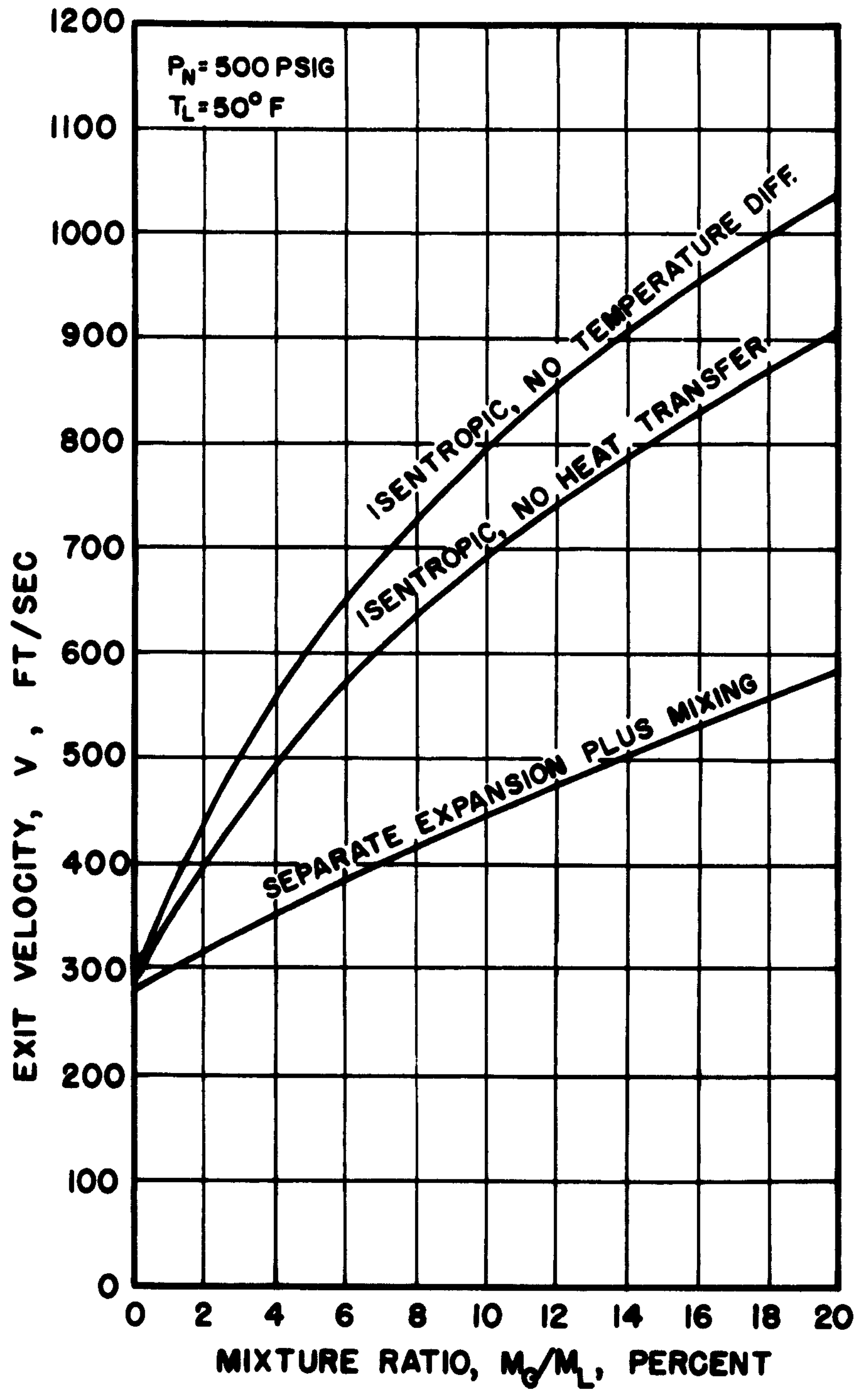


Fig. 7. Theoretical Exit Velocity as a Function of the Mixture Ratio for the Three Limiting Cases Using Air and Water

the highest exit velocity for a two-phase air and water mixture. The same result is applicable to the expansion, in the drive nozzle, of any non-cryogenic liquid where the initial temperatures of the two-phases are comparable (2). The criterion for preferring one type of expansion to another is the achievement of the maximum kinetic energy per unit mass of the drive liquid at the exit of the drive nozzle, with all of the other variables held constant.

Figure 7 shows that the case of separate expansion plus mixing is the least desirable process for transferring momentum to the drive liquid; it yields the smallest exit velocity for the two-phase mixture of the three types of expansion considered. For that reason the process involving separate expansion plus mixing was not investigated experimentally.

Another method of analysis of the two-phase flow in the drive nozzle that is independent of limiting cases was investigated. It was developed from simplified models of the dynamics of the flow of a single droplet of liquid, hereafter termed <u>droplet flow</u>, in an expanding gas (4). The analysis considers the effects of the drag of the gas on the liquid droplets, the secondary break-up of the liquid droplets, the geometric characteristics of the drive nozzle, and empirical relations for the thermal interactions between the gas and liquid phases. Experimental data were employed for taking into account the afore-mentioned effects in the analysis. The results obtained represent a semi-quantitative calculation of the variation of the phase velocities, droplet radius, and static pressure as functions of the axial distance along the drive nozzle.

None of the results predicted by the foregoing methods of analysis were in consistent agreement with the corresponding experimental results.

In general, the two analyses based on isentropic conditions were employed as a standard of performance for the experimental drive nozzles. The results of the non-isentropic analysis derived from the simplified models of droplet flow were satisfactory for correlating the indicated variations of droplet radius in the drive jet (4).

Figure 8 presents a sectional drawing of an experimental drive nozzle. All of the drive nozzles employed in the experimental investigation of drive nozzle performance incorporated a converging-diverging contour. In all, fifteen experimental drive nozzles were investigated.

The effects of changes in the following three principal design variables were studied: (a) length of the drive nozzle, (b) injector configuration, and (c) contour of the drive nozzle.

During the course of the experiments the length of the drive nozzle was varied from 2 in. to 9 in., and the flow rate from a few tenths of a pound per second to 20 lb per sec. The velocity recovery factor was determined with both non-cryogenic and cryogenic liquids. The results indicated that the short drive nozzles, with lengths of approximately 2.0 in., had the smallest velocity recovery factors; K_1 varied from 0.6 to 0.7. The low values of the velocity recovery factors were attributed to "slippage" of the droplets of the drive liquid in the drive gas because of the high acceleration transients (approximately 50,000 g's to 100,000 g's) occurring in the throat regions of the short drive nozzles.

In the case of the drive nozzles with lengths between 3.0 and 8.0 inches the velocity recovery factor varied between approximately 0.77 and 0.83, when the injector was designed to produce a uniform distribution of both phases over the face of the injectors, the gas and liquid being divided into 40 to 150 separate streams depending on the particular

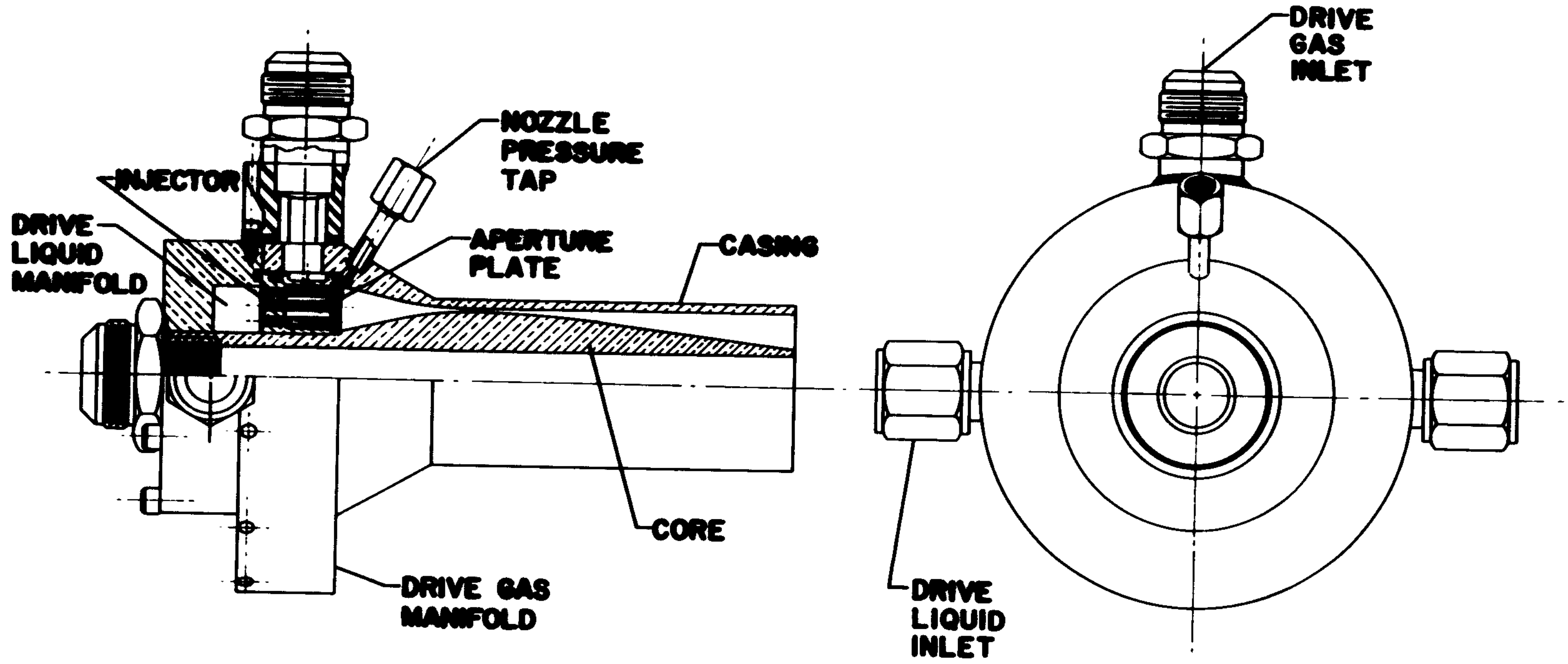


Fig. 8 Sectional Drawing of Drive Nozzle

configuration. With coaxial injection of the two-phases, that is, with a twin-fluid atomizer, it was found that the relative velocity between the higher velocity drive gas and the lower velocity drive liquid, at the injector face, had to exceed a certain minimum value for the velocity recovery factor to attain values in the range between 0.77 and 0.83. Apparently, when the relative velocity was smaller than the afore-mentioned minimum value, the streams of drive liquid were insufficiently atomized for achieving adequate mixing of the two phases. Accordingly, the momentum transfer from the gas to the drive liquid was small during the initial phase of the expansion process in the converging section of the nozzle, so that unusually large acceleration transients occurred in the throat region.

In some cases, it was possible to raise the value of velocity recovery factor of the drive nozzle by increasing the rate of contraction of its converging portion; that is, by shortening the converging portion for a fixed ratio of nozzle inlet area to throat area. As a consequence, the static pressure along the converging portion decreased more rapidly, and also the differential pressure between the injector face and the nozzle throat was increased, all other variables remaining constant. The aforementioned variation of static pressure indicated a more rapid decomposition of the streams of drive liquid and a subsequent increase in the amount of momentum transferred from the drive gas to the drive liquid in the converging section, thus increasing the velocity recovery factor.

In the expansion of a two-phase mixture of a cryogenic liquid and a gas the two fluids are essentially at the same temperature during the expansion; that is, they are in thermal equilibrium (2) (3). For a cryogenic liquid used in conjunction with a hot drive gas, thermal equilibrium during the expansion results in the lowest value of specific kinetic energy of the drive liquid at the exit of the drive nozzle. If the drive gas is condensible in the drive liquid, at the temperature of the drive liquid and the range of pressures occurring in the drive nozzle, condensation of the gas will occur during the expansion process. For example, the experiments with gaseous nitrogen as the drive gas showed that the nitrogen gas condensed when liquid nitrogen was used as the drive liquid. Thermal equilibrium during the expansion of a two-phase mixture of liquid nitrogen (-305 F), and gaseous helium (50F) gives a velocity recovery factor of approximately 75 per cent based on no heat transfer, which represents the most desirable type of expansion for achieving the maximum specific kinetic energy for the drive liquid. For the mixture of liquid nitrogen (-305 F) and gaseous nitrogen (50 F), the velocity recovery factor was 55 per cent based on no heat transfer. The reduction in the velocity recovery factor with liquid nitrogen as the drive liquid from 75 per cent with gaseous helium as the drive gas to 55 per cent for gaseous nitrogen as the drive gas, illustrates the undesirable influence of the condensation of the drive gas in the drive liquid.

The variation in the average size of the droplets of the drive liquid, in two or more drive jets was indicated by corresponding changes in the flow rate of drive liquid captured by a fractionating probe submerged in the drive jets (4). The average size of the droplets of the drive liquid in the drive jet was indicated as being a complicated function of the characteristics of the injector, the contour and length of the drive nozzle, and the mixture ratio. The experimental data obtained with the fractionating probe combined with the results of analytical calculations indicated that the surface mean diameters of the afore-mentioned droplets ranged between approximately 0.0005 in. and 0.005 in. The calculated surface mean diameters of the liquid droplets formed in the converging section of the drive nozzle by the twin-fluid atomizing injectors were approximately 0.040 in. Thus, the surface mean diameter of the droplets appeared to decrease by a factor of approximately 10 to 80 during the expansion of the two-phase mixture in the drive nozzle. A more complete discussion of the flow of a two-phase mixture in a drive nozzle is presented in references (1) (2) (3) (4).

It appears that the near optimum drive nozzle for a jet pump is one having a contour and length such that the velocity recovery factor would be a maximum when the injector produces liquid droplets in the converging section of the nozzle having a mean diameter of the same order of magnitude as that for the droplets in the drive jet. The optimum operating point for a drive nozzle will depend on the operating point and the overall design of the jet pump.

(c) The Mixer

The investigation of the characteristics of the mixer was divided into the following two principal phases: (1) the experimental determination of a satisfactory geometry for the mixer, and (2) the investigation of the process of momentum transfer from the drive liquid to the suction liquid.

The ideal mixer for a jet pump appears to be a channel wherein the drive liquid and the suction liquid mix in a free-stream flow. An actual mixer, however, because it is necessary that the suction liquid be distributed uniformly in the drive jet, requires a surface for accomplishing the mixing process. Any surface, of course, leads to frictional losses, and consequently, reduces the momentum recovery factor.

The design of an optimum surface-type mixer is not solely a function of its geometry, but one must consider simultaneously the characteristics of the drive nozzle, the geometry of mixer and the separator as a unit, and the method employed for injecting the suction liquid into the mixer. When all of the afore-mentioned factors are considered, the optimum geometry of a mixer surface is the one that maximizes the product of (a) the momentum recovery factor of the mixer, (b) the velocity recovery factor of the separator, and (c) the mass recovery factor evaluated at the diffuser.

Figure 9 illustrates five configurations of experimental mixers that were investigated, together with one which is deemed to be the optimum surface configuration for a mixer. Experiments by Elliott indicated that a 15 deg half-angle for the surface of the mixer gave the maximum recovery of momentum (1). Mixer types A, B, and C (in Fig. 9) were designed for operation in conjunction with an annular diffuser, whereas types D, E, and F for operation in conjunction with a conical diffuser. From the experimental results obtained with mixer Types A through E, it appears that the optimum geometry is one with a concave surface that curves toward a conical diffuser; the radius of curvature of the mixer surface would depend upon the configuration and dimensions of the drive jet, and also upon the method employed for injecting the suction liquid. No analytical description of the interdependence of the factors just described has been established. A conical mixer, identical to type D in Fig. 9, was employed in the investigation of the momentum transfer from the drive liquid to the suction liquid (5). No separator followed the mixer; instead an annular capture slot, the width of which could be

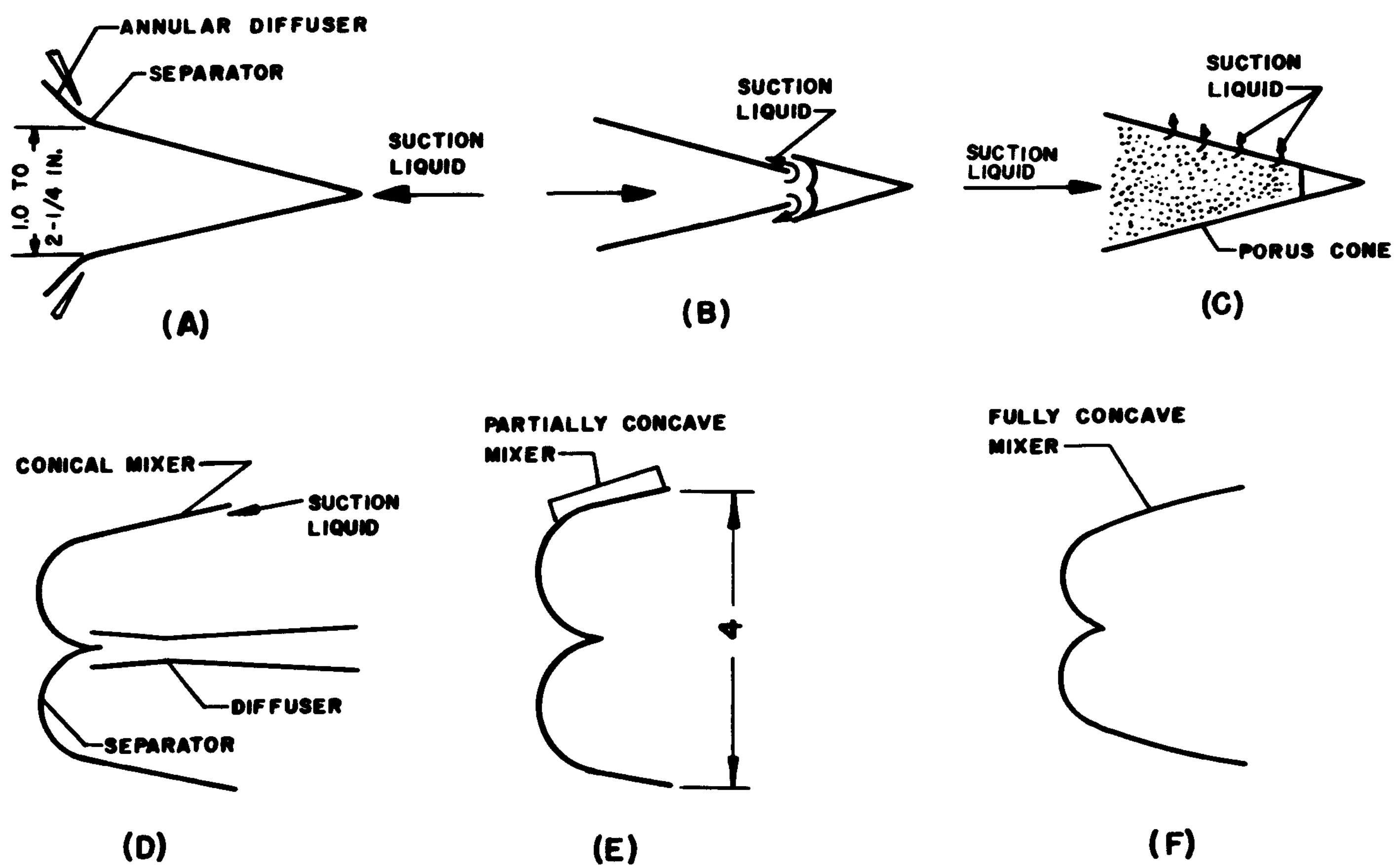


Fig. 9. Five Configurations of Experimental Mixers and the Proposed Optimum Configuration for a Surface Type Mixer

varied, was located at the exit of the mixer. The results of the experiments indicated that the mean velocity of the flow evaluated at the capture slot varied inversely with the $\dot{M}_S/\dot{M}_L$ ratio (where $\dot{M}_S$ is the suction liquid flow rate, and $\dot{M}_L$ the drive liquid flow rate), and directly with the nozzle mixture ratio $\dot{M}_G/\dot{M}_L$. The mass recovery of the combined flow rates of suction liquid and drive liquid was approximately 80 per cent for a capture slot width of 0.125 in.; the ratio of the volume of entrained drive gas to volume of liquid was in the ratio of 4 to 1.

Figure 10 presents representative curves of the local effective density and the local effective velocity of the flow on the mixer as a function of the width of the capture slot, as determined at the capture slot. It is seen that the density of the two-phase mixture decreases rapidly with s, until approximately s = 0.01 in. Thereafter, the density remained constant until s = 0.05, the limit of measurement. Figure 10 shows that the effective velocity of the two-phase mixture attains a maximum value at s = 0.008 in.; thereafter the velocity decreases with increasing values of s. There is reason to believe that the velocity of the flow within approximately 0.003 in. of the mixer surface is approximately equal to the injection velocity of the suction liquid (5).

A simplified flow model for the mixer is one comprising two principal regions (1) the free-stream drive jet wherein the drive liquid is suspended as droplets in the drive gas, and (2) the region within approximately 0.060 in. of the surface of the mixer where the liquid is assumed to be substantially the continuous phase but contains bubbles of entrained drive gas.

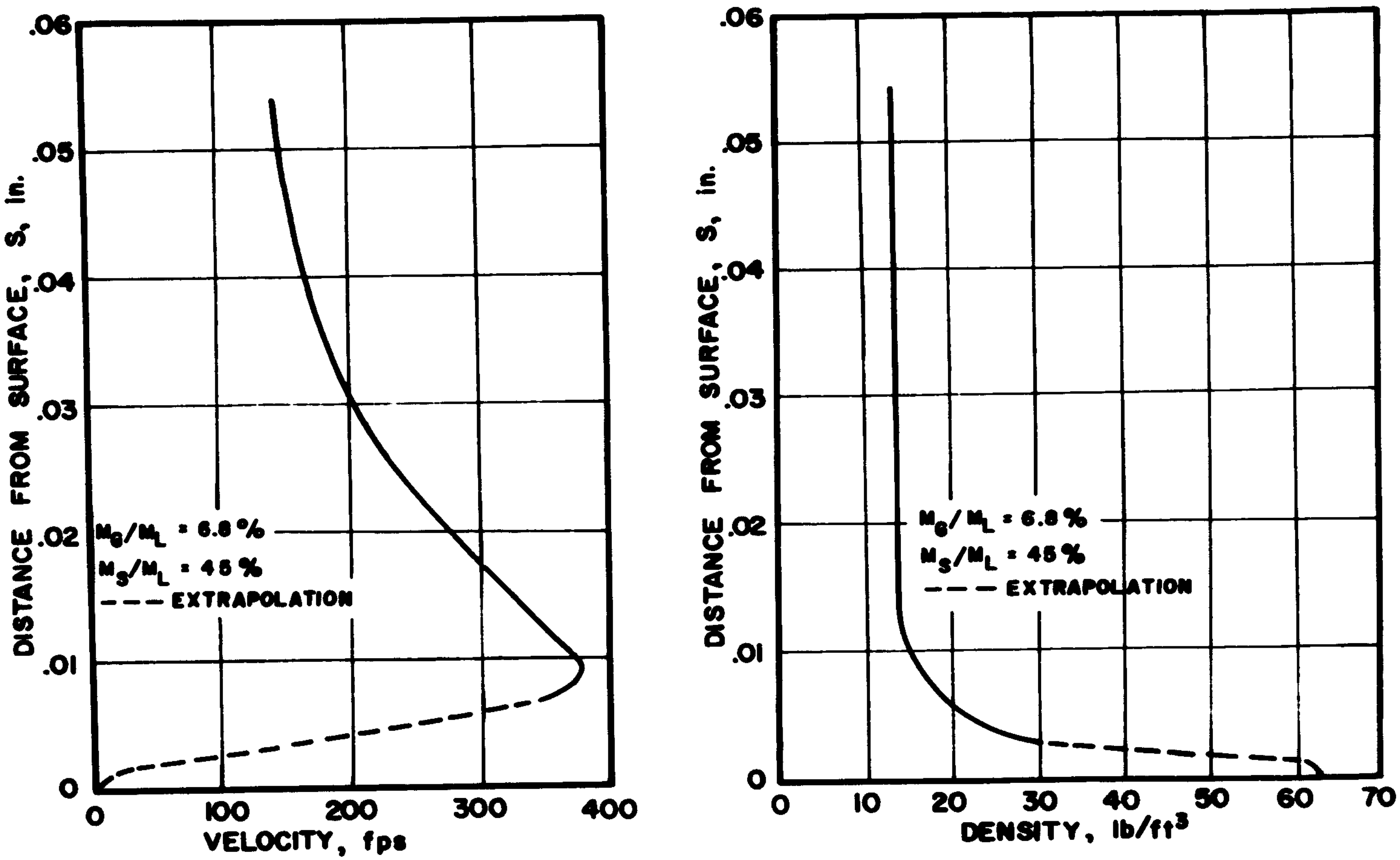


Fig. 10. Effective Velocity and Density Profiles at the Capture Slot

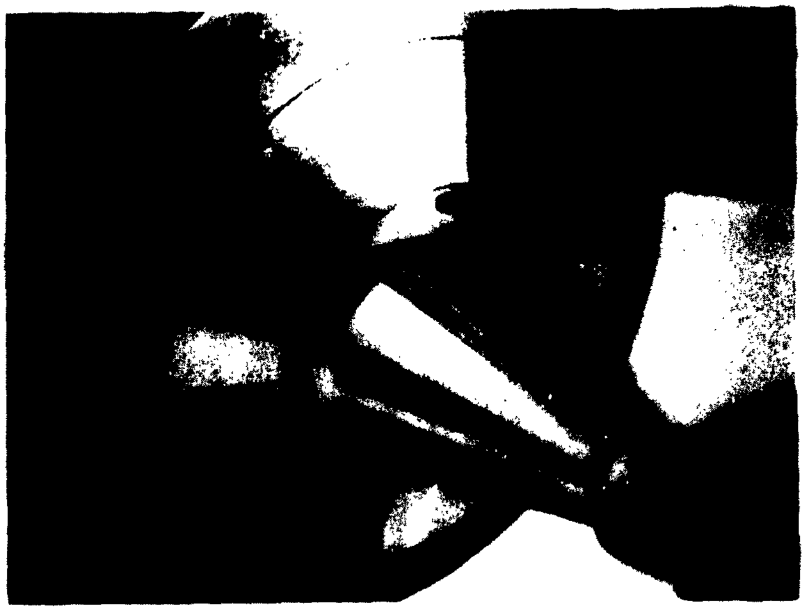
To predict the impingement characteristics of the droplets a simplified form of the mechanics of suspensions can be applied to it. The two-phase flow in the second region can be treated as a single fluid with position dependent properties. Moreover, since the local acoustic velocity at any point in the two-phase flow in the second region has a low value, approximately 75 fps (6) (7), the flow may be treated as a supersonic flow wherein the local Mach number varies from zero at the surface of the mixer to possibly eight at the point where the velocity of the two-phase flow is a maximum. In both of the afore-mentioned regions the flow is characterized by a high intensity of turbulence.

The momentum recovery factor for the mixer was determined experimentally for the mixer-types A through D, illustrated in Fig. 9. The momentum recovery was found to increase gradually from a value of approximately $K_2 = 0.7$ for no suction liquid flow to a value between 0.9 and 0.95 for ratios of suction liquid flow rate to drive liquid flow rate of approximately 0.5 and larger.

(d) The Separator

Figure 11 presents photographs of two types of experimental separators that were investigated, the separators were formed from a circular section of a toroidal surface. A radial acceleration, varying from approximately 10,000 g to 30,000 g was impressed on the two-phase mixture flowing through the separator inducing the separation of the entrained bubbles of drive gas from the two-phase mixture. Another function of the separator was the recovery of the liquid before the flow entered the diffuser.

An analytical criterion was established for defining the ideal separator. The criterion chosen was the time required for separating



(a)

(b)

Fig. 11 Two Configurations of Experimental Separators a single bubble, having a specified initial radius when adjacent to the surface of the separator, from a film of liquid of specified thickness, assuming the flow to be one-dimensional. The time of separation so calculated represented the minimum or ideal time.

The analytical relationships for calculating the ideal time of separation contained the following eight principal variables influencing the ideal separation process: (1) the velocity of the liquid film, (2) the density of the liquid film, (3) the initial diameter of the gas bubble, (4) the thickness of the liquid film, (5) the radius of curvature of the separator, (6) the drag coefficient of the gas bubble, (7) the assumed thermodynamic path taken by the gas in the bubble, for example, isothermal, isentropic, etc., and (8) the physical properties of the liquid composing the film, and the gas in the bubble.

Figure 11(a) is a photograph of the type of separator employed for determining the effective density and velocity profiles in the two-phase mixture flowing on a separator. Figure 12 presents the effective density as a function of the distance from the separator surface, and Fig. 13 presents the effective velocity as a function of the distance from the separator surface, both parameters being determined at the entrance to the diffuser. The particular separator employed in the experiments had a radius of curvature of 0.7 in. and an arc length of 30 deg. Comparing Figs. 12 and 13 with Fig. 10, it is seen that both the effective density profile and the effective velocity profile for the two-phase flow leaving the separator represent a continuation of the same profiles in the two-phase flow leaving the mixer. The density profiles in the flow leaving the separator can be readily extrapolated to a value of the density at

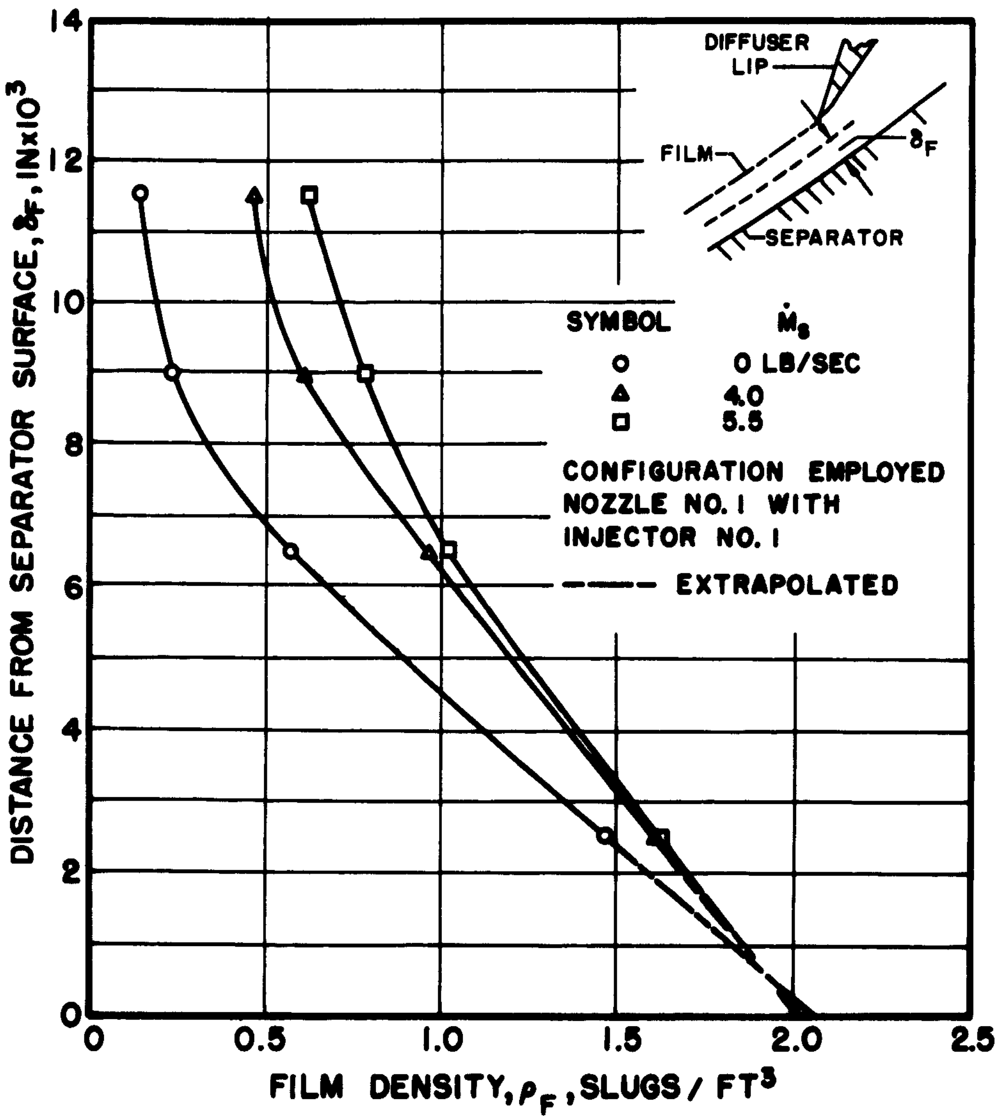


Fig. 12. Effective Density of the Film as a Function of the Distance from the Separator Surface

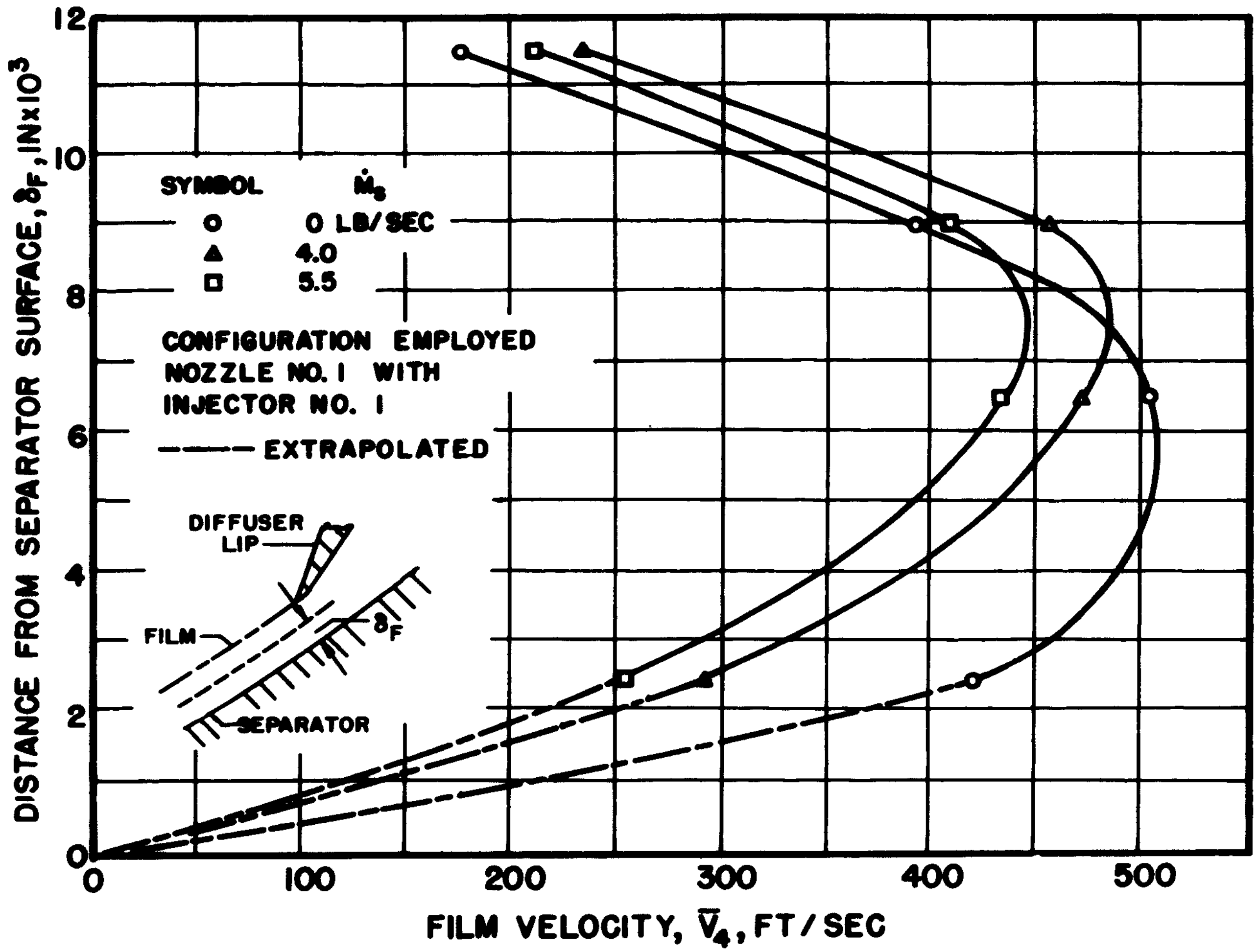


Fig. 13. Effective Velocity of the Film as a Function of the Distance from the Separator Surface

the surface of the separator, namely 2.0 slugs/ft^3, which is within a few per cent of the density of water, the liquid used in the experiments. Figure 12 shows that the mean density increased slightly with an increase in the suction liquid flow rate. The mean density of the two-phase flow did not vary linearly with the suction liquid flow rate; leaving the separator the mean density was approximately one-half the density of water.

The separator illustrated in Fig. 11(b) had a radius of curvature of 1.0 in. and an arc length of 165 deg. From the experiments with that separator, it was found that the density of the flow captured at the diffuser (in excess of 95 per cent of the total liquid flow) was approximately eight-tenths that of water. More complete separation was hindered by a continuous "thickening" of the flow as it curved inward toward the diffuser.

Substantially complete separation of the entrained drive gas from the liquid in a centrifugal separator of the type discussed herein would appear to be achievable if the effective thickness of the flow on the separator can be made small enough. Complete separation was observed in the first 5 deg of a centrifugal separator with a radius of curvature of 0.3 in. where the initial thickness of the flow was approximately 0.005 in.; flows of that thickness would be encountered in small jet pumps, however, and correspond to a drive liquid flow rate of the order of 1 lb per sec.

(e) The Diffuser

The investigation of the diffuser was directed toward determining the performance of diffusers with single-phase flow (water) and with two-phase flow (water and air); the inlet velocities were varied from 70 fps to 220 fps (8). The results of the investigation of a conical diverging diffuser, such as that normally employed for diffusing either an incompressible or subsonic gas flow, indicated that the problem of diffusing a two-phase flow, such as that leaving the separator of a gas-driven jet pump, is significantly different and more complex than that encountered in diffusing single phase flows in pipe-connected systems.

In a gas-driven jet pump the fluid to be diffused is a two-phase flow which enters the diffuser as a free jet. Because of the mixing and separation processes, there are large radial density and velocity gradients in the bubbly mixture entering the diffuser. Furthermore, if the bubbly mixture is treated as a continuum, for example as a dense gas in isothermal flow with a density approximately proportional to the pressure, the local acoustic velocity at any point in the fluid is quite low as noted earlier (approximately 75 fps). Consequently, if the mixture velocity is approximately 350 fps, as occurs in the jet pump, the flow has characteristics similar to that for a supersonic flow at a Mach number of approximately 5. The overall effect of the foregoing factors precludes the utilization of the conical diverging diffuser commonly employed for incompressible and subsonic compressible flows. It would appear that the diffusion of a two-phase flow requires a form of diffuser that can accommodate the supersonic characteristics of the flow.

The efficiencies of the diffusers employed in the experimental jet pumps varied between 0.35 and 0.65. Since no adequate physical or analytical description of the diffusion process involving a two-phase mixture of a gas and a liquid is available, it is not possible to state whether or not the diffuser efficiency can be increased to acceptable values, that is, between 0.8 and 0.9. No discussion of the problem of the diffusion of a bubbly mixture was found in the literature. A fuller understanding of the principal factors governing the diffusion of two-phase mixtures is required if satisfactory values of diffuser efficiency are to be achieved.

IV. EXPERIMENTAL GAS-DRIVEN JET PUMPS

Experiments were conducted with six experimental jet pumps. Two principal configurations were studied, one where the mixer was a full right circular cone and the diffuser was annular, and the other where the mixer was the internal surface of the base end of a right circular cone, and the diffuser was conical. The first four experimental jet pumps (designated Models A through D) had nominal discharge capacities between 0.5 lb/sec and 1.0 lb/sec. The experiments conducted with jet pumps A through D were of an exploratory nature, and the results obtained from those experiments were utilized in designing two larger jet pumps, the Model E and Model F.

Figure 14 presents a sectional drawing of the Model E jet pump. The effective length of the drive nozzle was 5.75 in. measured from the face of the injector to the nozzle exit. At a mixture ratio of 10.0 per cent, the drive liquid flow rate was 8.7 lb/sec. The suction liquid was injected onto the mixer from the hollow core of the drive nozzle, and the nominal suction liquid flow rate was 4.0 lb/sec. The separator was constructed with a 30 deg arc and a radius of curvature of 1.4 in., and the annular diffuser had a nominal throat thickness of 0.0100 in. with a 0.012 in. inlet approximately 0.002 in. upstream of the throat.

From the experiments with the Model E jet pump it was evident that the operation of that pump was deficient in three respects; (1) the diffuser had too low a mass recovery; (2) the separation of the entrained drive gas was inadequate; and (3) the diffuser had too low an efficiency (approximately 0.45). The major defect in the design in the Model E jet pump, however, was the separator. Its projected frontal area was too

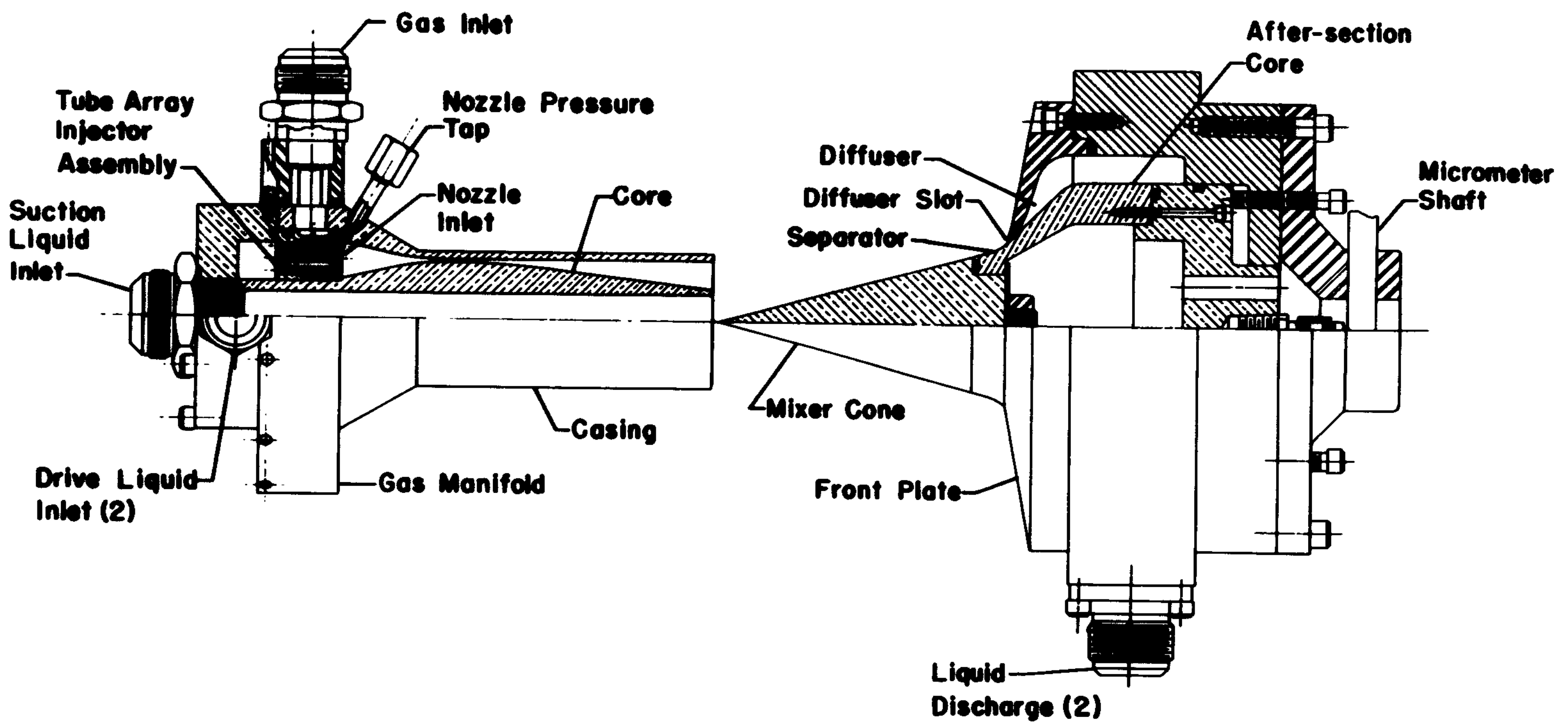


Fig. 14. Cross-Section of Model E Gas-Driven Jet Pump

small for achieving a satisfactory mass recovery, and its length of arc was too short for accomplishing a satisfactory separation of the drive gas from the liquid.

Figure 15 is a sectional drawing of the Model F jet pump. The effective length of the drive nozzle was 3.25 in. The drive liquid flow rate was 16.6 lb/sec at a mixture ratio of 7.0 per cent, and the suction liquid was injected onto the mixer through an annular slot located close to the exit of the drive nozzle; the nominal flow rate of suction liquid was 8.0 lb/sec. The separator had an arc length of 165 deg with a radius of curvature of 0.9 in. In one experimental version the radius of curvature was 1.0 in. and the first 30 deg of the separator also served as a portion of the mixer. Both a diverging diffuser, as shown in Fig. 15, and a converging-diverging diffuser were employed in the Model F jet pump.

Table 1 presents the operating characteristics of the Model E jet pump, and the two configurations of the Model F jet pump. Of particular interest are the values of the volumetric air entrainment, the maximum discharge pressure, and the mass recovery at the maximum discharge pressure when the experimental jet pumps were operating at their nominal pumping capacity which was $\dot{M}_S/\dot{M}_L = 0.5$.

The experiments showed that the volumetric air entrainment of the bubbly mixture entering the diffuser of the Model F-II jet pump, which had a 165 deg separator arc, was less than one-half of that for the Model E jet pump; the latter had a 30 deg separator arc.

The maximum discharge pressure achieved experimentally, at the nominal suction liquid flow rate, was 450 psi for the Model F-I and 400 psi for the Model F-II jet pumps, respectively. The lower value

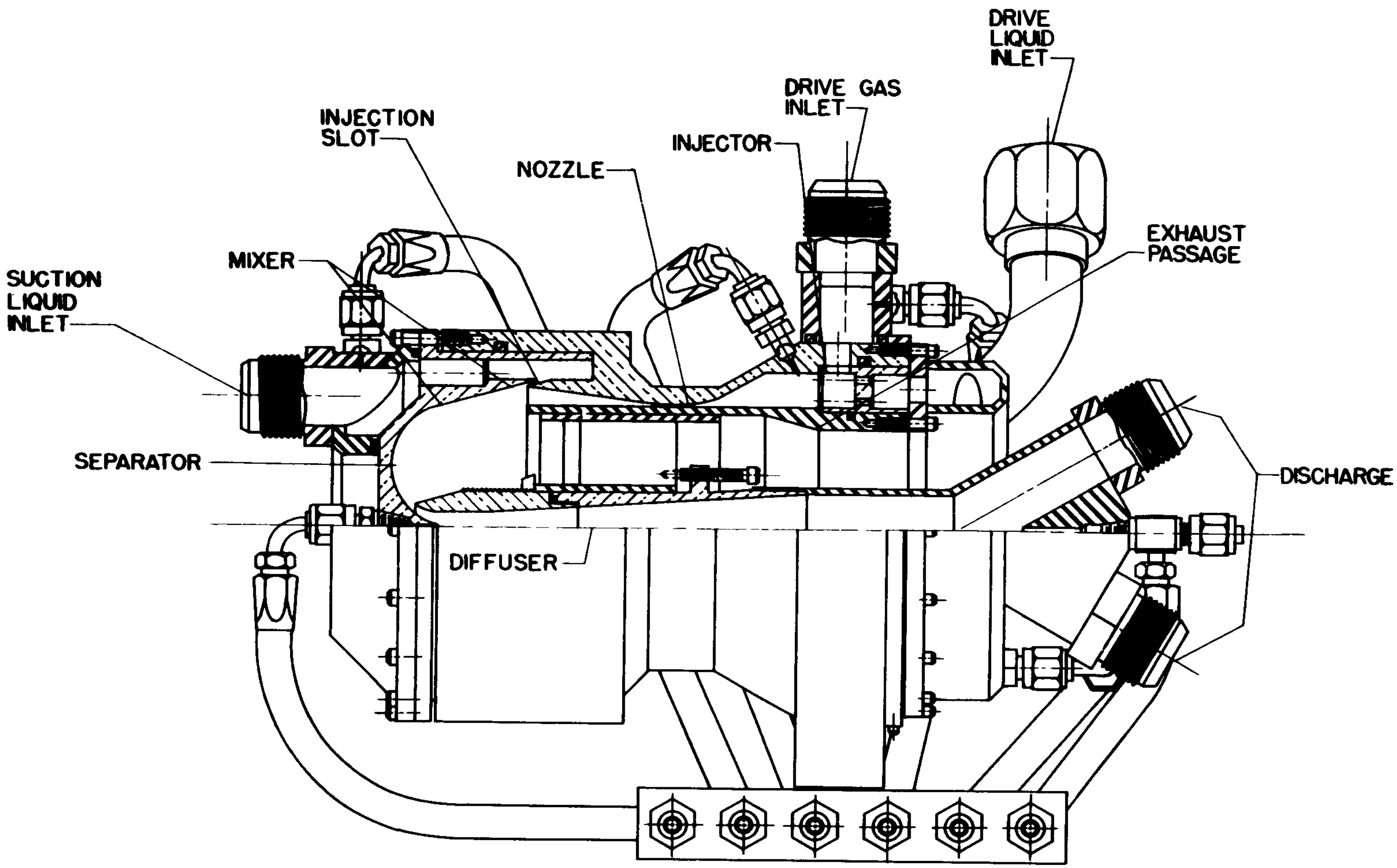


FIG. 15 CROSS-SECTION OF MODEL F GAS-DRIVEN JET PUMP

Table 1 The Operating Characteristics of the Model E Jet Pump and the Model F Jet Pump

Parameter at M_G/M_L* = 7%	Model E	Model F-I**	Model F-II
Mass recovery at the diffuser, $M_S/M_L = 0$, $P_D \cong 100$ psi, per cent	92	98	100
Mass recovery at the diffuser, $M_S/M_L = 0.5$, $P_D \cong 100$ psi, per cent	64	65	97
Volumetric air entrainment (Q_a/Q_T) at diffuser, $M_S/M_L = 0.5$, $P_D \cong 100$ psi, per cent	50 (30 deg separator arc)	--	25 (165 deg separator arc)
Maximum discharge pressure, $M_S/M_L = 0.5$, psig	360	450	400
Mass recovery at maximum discharge pressure, $M_S/M_L = 0.5$, per cent	less than 55	less than 65	95

* M_G = the mass flow rate of drive gas; M_L = the mass flow rate of drive liquid; M_S = the mass flow rate of suction liquid; P_D = the diffuser discharge pressure; (Q_a/Q_T) = the ratio of the volume of entrained air to the total volume evaluated at the diffuser entrance.

** Model F-1 employed conical mixer and diverging diffuser; Model F-II employed partially concave mixer and converging-diverging diffuser.

obtained with the Model F-II jet pump, is attributed, in part, to the design of the mixer which was not the optimum design for achieving the maximum momentum recovery. Both of the diffusers employed in the Model F "stalled out" abruptly with a sudden decrease in mass recovery at a pressure approximately 10 psi to 20 psi higher than the maximum discharge pressure presented in Table 1. The afore-mentioned stalling is believed to be a characteristic associated with the diffusion of the supersonic two-phase flow.

A satisfactory mass recovery factor (95%) was obtained with the Model F-II jet pump at the nominal suction liquid flow rate and the maximum discharge pressure. The low mass recovery factors obtained for the Model E and Model F-I jet pumps (less than 55 per cent and 65 per cent, respectively) are attributed to the inadequacies of the diffuser. In both jet pumps the diffusers were essentially designs that are applicable to either incompressible of subsonic flows. They caused a discontinuity to form in the two-phase flow as it approached the diffuser entrance, in a manner similar to a compression shock.

The experimental values of the momentum recovery factor of the drive nozzle, the mixer, and the separator were sufficiently large to indicate that a practical gas-driven jet pump is possible for achieving a net flow of discharged liquid at a pressure between 600 psi and 650 psi provided the diffuser efficiency can be raised to approximately 0.8. To realize a practical jet pump, either more accurate knowledge is needed regarding the diffusion of two-phase mixtures, or a more satisfactory method must be developed for separating the entrained drive gas from the mixture entering the diffuser.

V. CONCLUSIONS AND RECOMMENDATIONS

The conclusions and recommendations presented herein are based primarily on the analytical and experimental results obtained from the investigations discussed in this report. The conclusions presented apply to the feasibility and limitations of the gas-driven jet pump, and to the components of such a jet pump. More details regarding the technical description of the jet pump and its components are presented in the references (1) (2) (3) (4) (5) (8).

(a) Conclusions for the Gas-Driven Jet Pump System

1. The analytical and experimental investigations of the gas-driven jet pump revealed no fundamental thermodynamical or fluid mechanical reason which would preclude the pumping of a net flow rate of a liquid to a discharge pressure in excess of the nozzle pressure.
2. The results of the analytical investigation indicated that values of the gas-consumption between 5.0 per cent and 10.0 per cent should be attainable for gas-driven jet pumps in which the momentum recovery factor of each component is approximately 0.9.
3. The gas-driven jet pump does not appear applicable to pumping cryogenic propellants or propellants with uncommonly high vapor pressures because the high vapor pressures cause an excessive loss of propellant in the form of vapor.
4. The operating characteristics of a gas-driven jet pump are strongly influenced by the shape and mechanical design of its components.

(b) Conclusions for the Components of a Gas-Driven Jet Pump

1. Based on both analytical* and experimental results the velocity recovery factor of the drive nozzle appears to be limited to a value below 0.9 for a drive nozzle employing an injector having a differential pressure of injection which is compatible with the system requirements.
2. Condensibility of the drive gas in the drive liquid is a definite impediment to the attainment of a high velocity for the two-phase mixture leaving the drive nozzle.
3. It is possible to mix the drive liquid with the suction liquid, in the mixer, and achieve a momentum recovery in excess of 90 per cent.
4. Experiments showed that the volume of the entrained drive gas in the two-phase mixture entering a properly designed centrifugal separator could be reduced by as much as 95 per cent in flowing through that separator.
5. A diffuser for diffusing two-phase mixtures is needed if there is incomplete separation of the entrained drive gas in the mixture entering the diffuser.

(c) Recommendations

It is recommended that the following research programs be initiated.

1. A theoretical and experimental investigation for determining the variables and fluid mechanical processes that limit the velocity recovery factor of a two-dimensional drive nozzle.

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* Personal communication with Dr. D. G. Elliott, Jet Propulsion Laboratory, NASA.

2. A theoretical and experimental investigation of the process of the separation of entrained gas in a liquid flowing in a two-dimensional centrifugal separator.
3. Theoretical and experimental investigation of the process of the diffusion of a two-phase mixture of a liquid and a gas for different amounts of gas entrainment.
4. An investigation for improving the instrumentation techniques for measuring the pertinent variables of two-phase mixture of a gas and a liquid.

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